

Investigation of activated sludge pumping through a case study

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ABSTRACT

The energy consumption of the drinking water and wastewater technologies is growing, and gaps remain in the field of estimation of hydrodynamic losses. The pumping of recirculated sludge in the comprehensive activated sludge technology exposes 1.5–3% of the electric energy consumption of a wastewater treatment plant (WWT). At several technological points of the whole activated sludge process, changes in the total suspended solids (TSS) content of the activated sludge can occur. These changes cause sedimentation problems and also modify the rheological properties of the fluid. In our work, we determined the losses and performance curves of a simple hydrodynamic system carrying three different, real activated sludges based on computational fluid dynamics (CFD) calculations and literature data. We have shown that even a change in the sludge TSS content can have negative effect on the specific energy consumption of the system.

INTRODUCTION

In recent years, energy-efficient thinking has been appreciated in all areas of industry. Water supply and wastewater

services are also of significant importance in terms of both energy consumption and environmental emissions (Longo, S. et al. (2016); Wakeel, M. et al. (2016); Haaz, E. et al. (2019)).

The energy consumption of the wastewater treatment (WWT) sections varies in a wide range depending on many parameters, such as the size and location of the plant, water contaminations, or the treatment process type. The WWT process consists of several sludge treatment steps with different energy requirements. In general, the primary treatment requires less energy than the other stages. The energy consumption of the secondary treatment is significantly higher and depends on technology and contamination. The tertiary treatment is even more energy-intensive step depending on the treatment standard of the given country. Based on Wakeel, M et al. (2016), the average specific energy consumption of WWT is between 0.25 to 0.67 kWh/m³.

The conventional type of secondary treatment is the *activated sludge process*. The specific energy consumption of the needed sludge recirculation pumps were reported to be between 0.0047 to 0.01 kWh/m³ (Longo, S. et al.), and in average of 0.011 kWh/m³ (Plappally, A.K. &

Lienhard V.J.H. (2012)). This represents 1.5– 3.5% of the total electric energy consumption of the whole WWT plant (Longo, S. et al. (2016)).

For designing and operating these technologies, it is essential to examine the rheological properties of the sludge, which is known as non-Newtonian fluid. For the rheological modelling of activated sludge, the literature suggests the Ostwald (power-law), the Bingham, and the Herschel-Bulkley models (Ratkovich, N. (2013); Seyssiecq, I., Karrabi, M. & Roche, N. (2015)). Material properties of the fluid vary depending on the total suspended solids (TSS) (Guibaud, G. et al. (2004)), temperature, and material structure, even at several technological points of the whole activated sludge process (Baroutian, S., Eshtiaghi, N. & Gapes, D.J. (2013); Füreder, K. et al. (2018)). Any adverse changes in the fluid structure may cause settling problems, and modify rheology of

the fluid as well (Bakos, V., Kiss, B. & Jobbágy, A. (2016)).

The performance curve of a pump and the characteristic curve of a pipeline system are usually plotted as the head (H [m]) as the function of the volume flow rate (Q [m^3/h]). The intersection of the two curves is the operation point. The hydrodynamic losses of the straight pipes and pipe elements depend on the rheology. In the case of a non-Newtonian fluid, the characteristic curve of a pipeline-system differs from those delivering Newtonian fluids (Chhabra, R.P. & Richardson, J.F. (2008)). A remarkable deration of the head appears on the non-Newtonian performance pump curve at the lower range of the volume flow rate as well (Walker, C.I. & Goulas, A. (1984)). This could indicate a second (or a third) intersection of the curves, as **Fig. 1** shows based on (Graham, L.J.W. et al. (2009)).

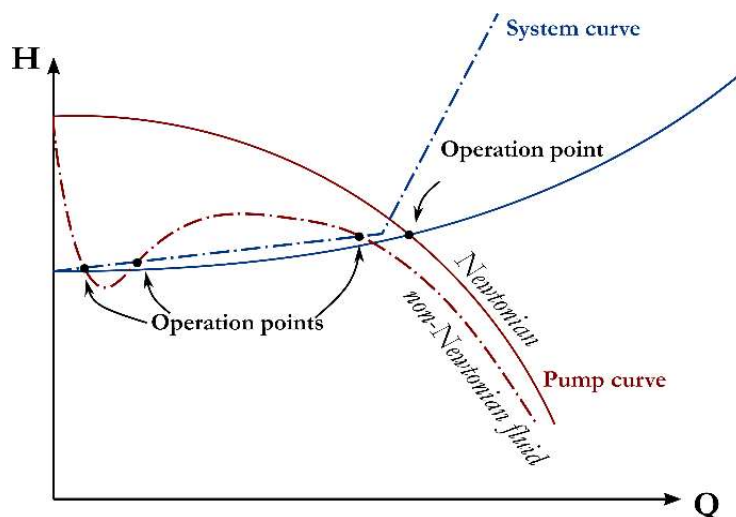


Fig. 1. System curve (blue lines) and performance curve of centrifugal pump (red lines) for Newtonian (solid lines) and non-Newtonian (chain lines) fluids with the operation points, based on Graham, L.J.W. et al. (2009)

A large and growing body of literature has investigated the friction factor in straight pipes and the loss coefficients of pipe elements in the case of non-Newtonian fluids. Garcia and Steffe (Garcia, E.J., & Steffe, J.F. (1986)), Turian et al. (Turian, R.M. et al. (1998A)), Telis-Romero, Telis and Yamashita (Telis-Romero, J., Telis, V.R.N. & Yamashita, F. (1999)), and Haldenwang et al. (Haldenwang, R. et al. (2012)) investigated the friction losses in straight pipes. Non-Newtonian fluid through valves was presented by Pienaar et al. (Pienaar, V.G. et al. (2004)), through bends, valves, and Venturi-meters by Turian et al. (Turian, R.M. et al. (1998B)), and through valves and fittings by Polizelli et al. (Polizelli, M.A. et al. (2003)). The computational fluid dynamics (CFD) has proven to be a reliable tool for the estimation of pipeline system losses (Csizmadia, P. & Hős, Cs. (2014)).

Publications that concentrate on non-Newtonian losses more frequently apply new definitions of friction factor (Garcia, E.J., & Steffe, J.F. (1986)) or Reynolds number (Haldenwang, R. et al. (2012)). Madlener et al. defined the generalized Reynolds number for power-law, Bingham plastic, and Herschel-Bulkley fluids (Madlener, K., Frey, B., Ciezki, H.K. (2009)). It has been demonstrated by the authors previously that among the many existing definitions, the generalized Reynolds number (Re_{gen}) arranges the friction factor values appropriately (Csizmadia, P. & Till, S. (2018)).

The literature on centrifugal pump delivering non-Newtonian fluid suggests the usage of the standard 12.1-12.6 (2005) formed by the Hydraulic Institute in conjunction with the American National Standard Institute (ANSI/HI) (Addie, G.R.,

Roudnev, A.S. & Sellgren, A. (2007)). In the calculations described in the standard, it is necessary to specify the viscosity of the fluid, which is not evident for non-Newtonian materials (Heywood, N. & Slatter, P. (2019)). Different methods exist in the literature: the Walker and Goulas method (Walker, C.I. & Goulas, A. (1984)), the Pullum et al. method (Graham, L.J.W. et al. (2009)), and the method developed by Sery et al. (Heywood, N. & Slatter, P. (2019)).

As it was shown, a considerable amount of literature has been published on pumping of non-Newtonian fluids. These studies, to date, have not yet determined the effect of the increasing TSS concentration on pumping efficiency and operation point of the system. The purpose of our investigation was to demonstrate its impact through this case study.

MATERIALS AND METHODS

Rheological modelling

Guibaud et al. (Guibaud, G. et al. (2004)) published measurement data of activated sludge from WWTs' aeration tanks with different TSS contents (7.4 g/l; 6.2 g/l; 3.6 g/l). The conditions of the measurement and the fit of the non-Newtonian models were detailed in our previous study (Csizmadia, P. & Till, S. (2018)). For rheological modelling, we used the Bingham model, in which there is a linear relationship between the τ [Pa] shear stress and the $\dot{\gamma}$ [1/s] shear rate, as $\tau = \tau_0 + \mu_B \dot{\gamma}$, where τ_0 [Pa] is the yield stress and μ_B [Pas] is the dynamic viscosity. The fitting on the measured value was carried out by the least squares method using MatLab software. Table 1

shows the exact rheological parameters and the fit rheograms of the three sludges supplemented by that of water.

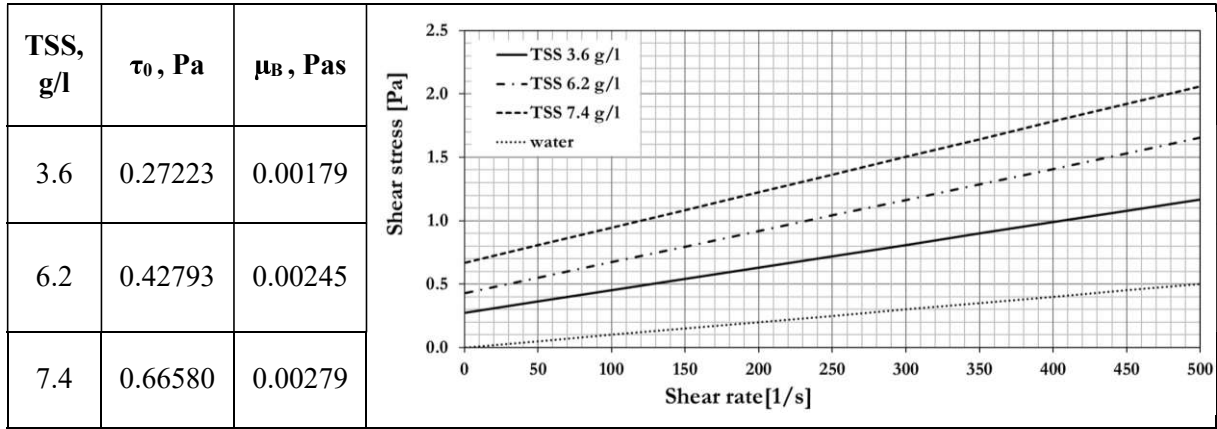


Table 1. The Bingham plastic rheological parameters of the investigated sludges and the fit rheograms according to Guibaud et al. (Guibaud, G. et al. (2004)) and Csizmadia and Till (Csizmadia, P. & Till, S. (2018))

The modelled hydrodynamic system

In our case study, we designed a fictive hydrodynamic system (see in **Fig. 2**) with the static head of $H_{st}=5\text{ m}$. Our case system consisted of a 30 m long straight pipe with eight elbows on it. The inner pipe diameter was $D=0.1\text{ m}$; the relative

radius of curvature of all the bends was the same as $R/D=1$, where $R\text{ [m]}$ is the centreline radius of the curvature. We assumed a hydraulically smooth pipeline. The selected centrifugal pump operated at revolution number of $n=1\,420\text{ 1/min}$, the impeller diameter of the pump was $D_{imp}=0.138\text{ m}$.

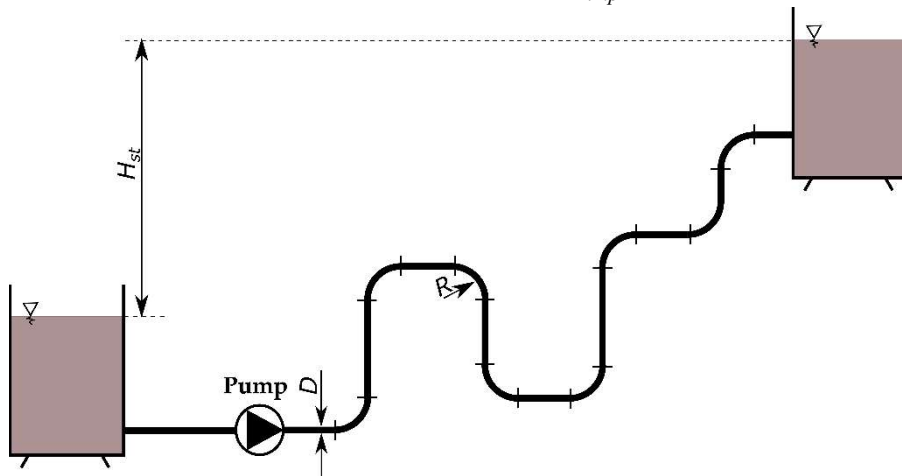


Fig. 2. The modelled hydrodynamic system

Estimation of the pressure losses of the pipeline system

Losses of straight pipe and pipe elements were determined based on our previous work, using computational fluid dynamic (CFD) simulations, for details see Csizmadia, P. & Hős, Cs. (2014); Csizmadia, P. & Till, S. (2018); and Csizmadia, P. & Hős, Cs. (2013). The geometry of the elbow was built as a fully structured 3D numerical O-grid type mesh (generated in ICEM CFD), including 1.2 million cells, see in **Fig. 3a**. The pipe section was modelled as a no-slip, hydraulically smooth wall. Steady-state computations were performed with the commercial ANSYS-CFX software (Ansys, C. (2009)), and a high-resolution spatial scheme was used for all equations. The actual turbulence model was the SST (shear stress transport) turbulence model. Additional straight pipes with a length of $50D$ to the upstream and another one with a length of $10D$ to the downstream sides were added for allowing proper boundary conditions. At the inlet the fully developed turbulent velocity profile, and at the outlet, the average static pressure was prescribed as boundary conditions. **Fig. 3b** shows the velocity streamlines in terms of TSS 3.6 g/l and a mean velocity of 1 m/s.

Using the generalized Reynolds number (Madlener, K., Frey, B., Ciezki, H.K. (2009)), the friction factor of the pipe was estimated using the Darcy equation ($f_D=64/Re_{gen}$) for laminar flow conditions and the Blasius equation ($f_B=0.316/Re_{gen}^{0.25}$) for turbulent flows (Csizmadia, P. & Till, S. (2018)), the results are shown in **Fig. 3c**. The loss coefficients of the elbows were calculated from the total pressure loss caused by the pipe element. Our CFD calculations were performed on all three rheology models in the mean velocity range of $v=0-2$ m/s. In **Fig. 3d** can be seen, that the loss coefficients were determined as the function of the generalized Reynolds number from the simulation results.

Estimation of the pump curve

The calculation of a centrifugal pump performance curve, which is delivering high viscous Newtonian flow, has been well established. This so-called Hydraulic Institute method assumes that the viscosity of the fluid is constant, see (Addie, G.R., Roudnev, A.S. & Sellgren, A. (2007)). For these calculations, a characteristic parameter (B) has to be assessed from the data of the best efficiency point (BEP) with water:

$$B = 16.5 \frac{v^{0.5}(H_{BEP-water})^{0.0625}}{(Q_{BEP-water})^{0.375}n^{0.25}} \quad (1)$$

where v [$cSt=10^{-6}m^2/s$] is the kinematic viscosity of the fluid; H [m] is the head; Q [m^3/h] is the volume flow rate, and n [rpm] is the revolution number, see e.g. (Addie, G.R., Roudnev, A.S. & Sellgren, A. (2007)). If $B > 40$, the fluid properties do not change the performance curve.

The modified values of volume flow rate ($Q'=C_q \cdot Q_{water}$), head ($H'=C_H \cdot H_{water}$) and efficiency ($\eta'=C_\eta \cdot \eta_{water}$) are derived from those with water using the following multipliers based on e.g. (Kalombo, J.-J.N. (2014)):

$$C_Q = 2.71^{-0.165 (\log B)^{3.15}} \quad (2)$$

$$C_H = 1 - \left[(1 - C_Q) \left(\frac{Q_{\text{water}}}{Q_{\text{BEP-water}}} \right)^{0.75} \right] \quad (3)$$

$$C_\eta = B^{-(0.0547 B^{0.69})}, \text{ for } 1 < B < 40 \quad (4)$$

$$C_\eta = \frac{1 - \left[(1 - \eta_{\text{BEP-water}}) \left(\frac{v}{v_{\text{water}}} \right)^{0.07} \right]}{\eta_{\text{BEP-water}}}, \text{ for } B < 1 \quad (5)$$

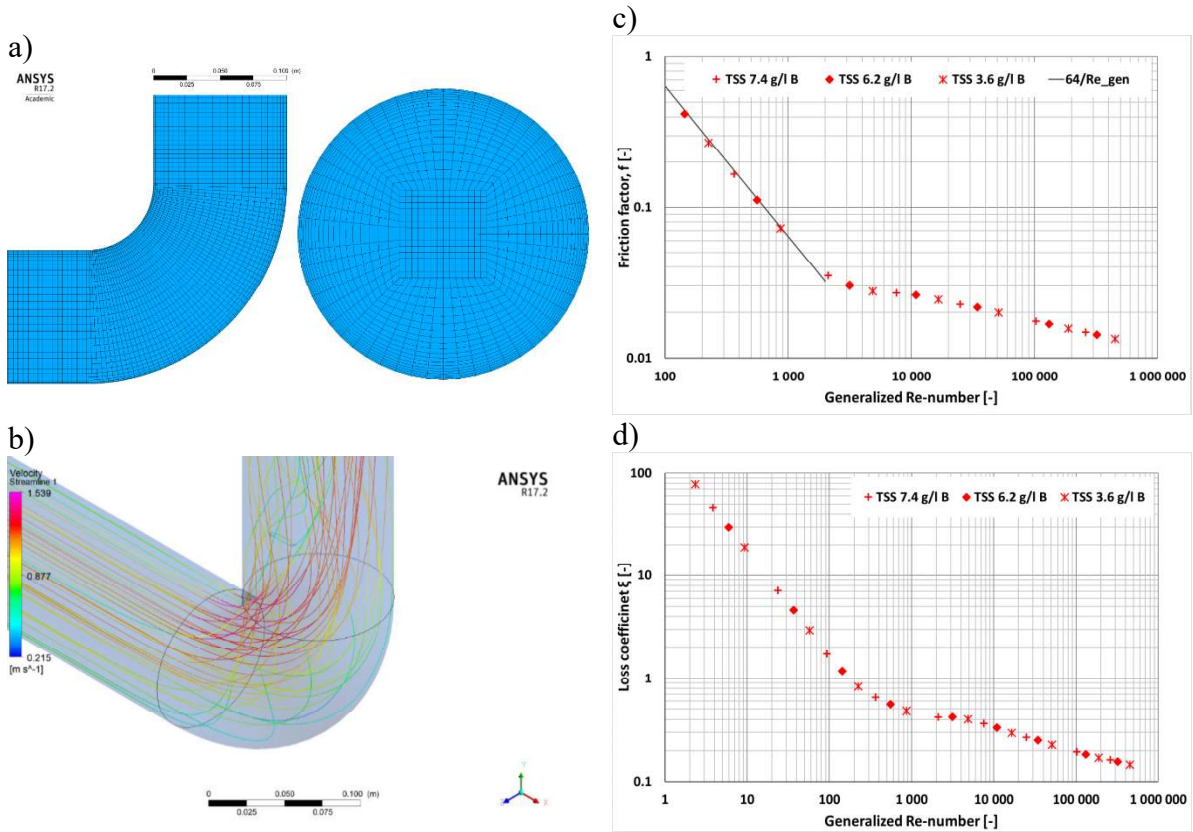


Fig. 3. The details of the CFD simulations based on Csizmadia, P. & Till, S. (2018)
 a) the numerical mesh; b) velocity streamlines plot at TSS 3.6 g/l and $v=1$ m/s; c) the calculated friction factors as the function of the generalized Re-number; d) the calculated loss coefficients as the function of the generalized Re-number

In case of non-Newtonian fluids, the kinematic viscosity ($\nu = \mu/\rho [m^2/s]$, where $\mu [Pas]$ is the dynamic viscosity, and $\rho [kg/m^3]$ is the density) is not a constant, but depends on the shear rate. This parameter appears in the Eq. (1) and thus indirectly affects all correction factors. Based on the Hydraulic Institute method, the pump curve for non-Newtonian fluids can be calculated only with some modifications. Graham et

al. recommend establishing a representative shear rate; using that, we can determine the representative viscosity for the calculations (Graham, L.J.W. et al. (2009)).

For laminar flow conditions, the equivalent ‘pipe’ diameter (D_{eq}) and from that, the equivalent velocity (v_{eq}) through the ‘pipe’ is calculated:

$$v_{eq} = \frac{4Q}{\pi D_{eq}^2} = \frac{4Q}{\pi \left(\frac{4w\pi D_{imp}}{2(\pi D_{imp} + w)} \right)^2} \quad (6)$$

where $D_{imp} [m]$ is the impeller diameter, and $w [l]$ is a characteristic dimension of the impeller, see (Graham, L.J.W. et al. (2009)). Some researchers (Graham, L.J.W. et al. (2009); Furlan, J., Visintainer, R., Sellgren, A. (2016); and Aldi, N. et al. (2017)) have attempted to specify the parameter w , but Kalombo et al. (Kalombo, J.-J.N. (2014))

ascertained that it must remain an empirical approach for it. In our study, following the suggestion of Graham et al. (Graham, L.J.W. et al. (2009)), we estimated w , as the 25% of the impeller diameter.

The shear rate is obtained from the Rabinowitsch-Mooney equation:

$$\dot{\gamma} = \left(\frac{3n' + 1}{4n'} \right) \frac{8v_{eq}}{D_{eq}} \quad (7)$$

where n' is the local gradient of the curve $d \ln(\tau) / d \ln(8v_{eq}/D_{eq})$. In the case of turbulent flow, the typical viscosity is

calculated from the rheogram at the shear rate of $\dot{\gamma} = 4\,000\,1/s$ (Graham, L.J.W. et al. (2009)).

RESULTS

The rheological changes caused by increasing tss content resulted in a barely noticeable change in the characteristic curves of the case system. They generated only a slight increase in the head values around the operation point. Several ‘curvatures’ appeared on the system curve around the laminar-turbulent transition. in

contrast, the pump curve showed a significant decrease in the head in the range of low volume flow rates, see **Fig. 4**. These results are following the curves given in the study of Walker and Goulas (Walker, C.I. & Goulas, A. (1984)) and Graham et al. (Graham, L.J.W. et al. (2009)).

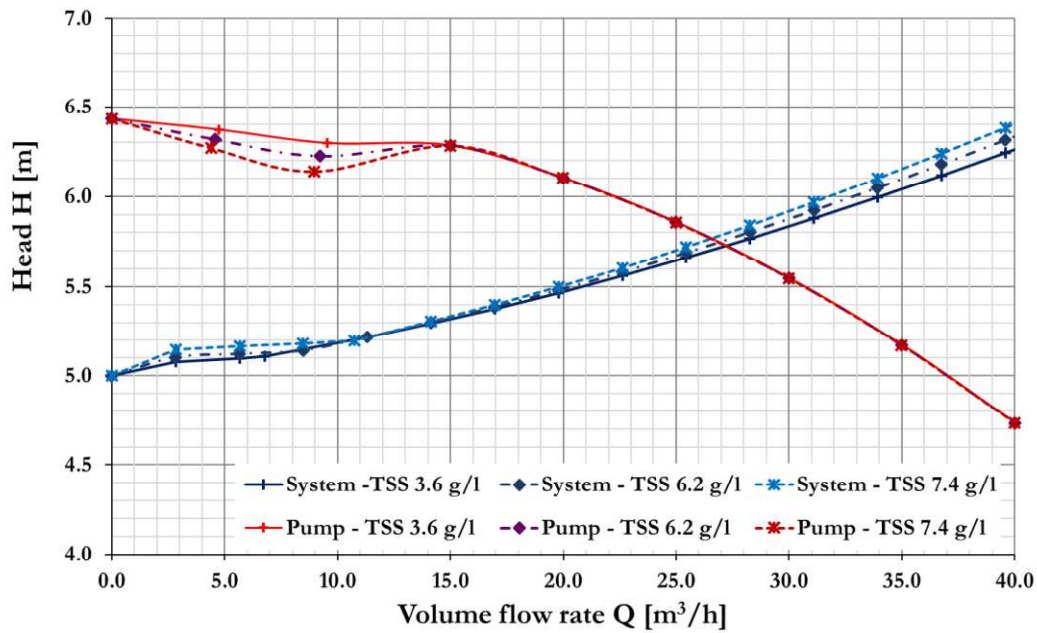


Fig. 4. The system curve (blue lines) and the pump performance curve (red lines) in case of TSS 3.6 g/l (solid lines), TSS 6.2 g/l (chain lines) and TSS 7.4 g/l (dashed lines)

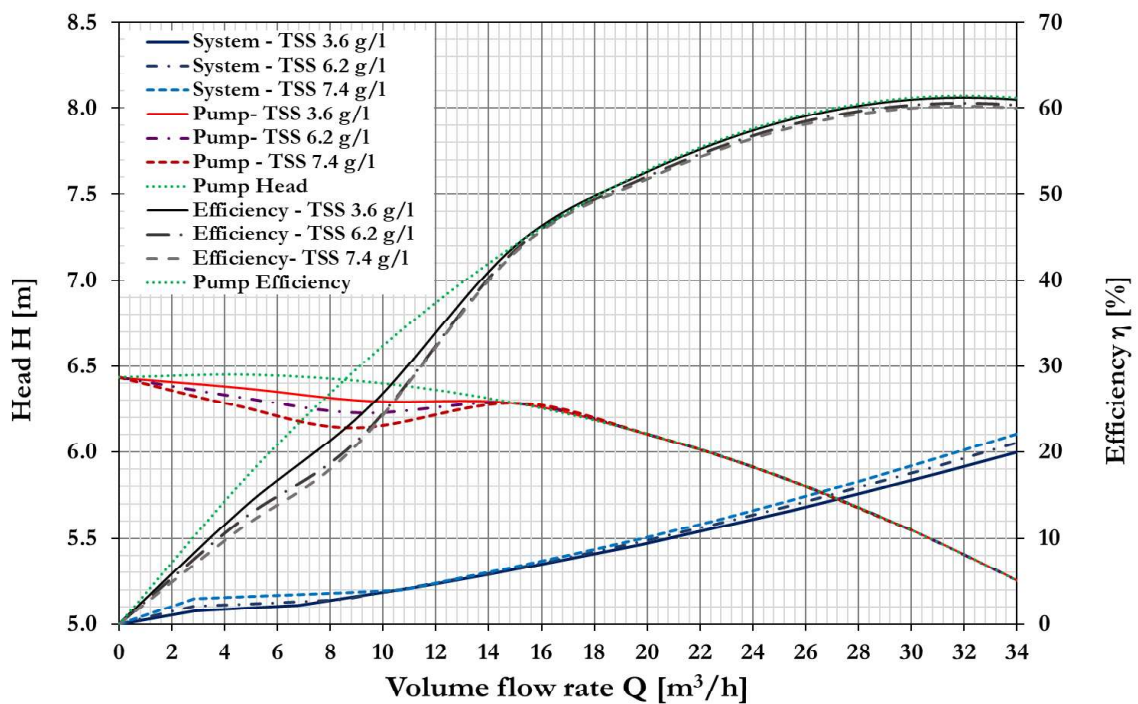


Fig. 5. Operation points and efficiency curves in case of TSS 3.6 g/l (solid lines), TSS 6.2 g/l (chain lines) and TSS 7.4 g/l (dashed lines); original head and efficiency curves of the pump (dotted green line)

Although there is no significant shift in the operating point of the system, the deterioration in pump efficiency is worth mentioning, as shown in **Fig. 5**. In the case of TSS content of 3.6 g/l, the pump operated near the point of maximum efficiency; and this fact did not change by the other two fluids, but the efficiency value itself slightly decreased. It can be

seen from the data in Table 2 that in the case of TSS 6.2 g/l, the efficiency reduced to 59%; and to 58.4% in the case of the TSS 7.4 g/l. The specific electric energy consumption of the pump was also estimated. The results showed that the changes in TSS content could cause an up to 3% increase in pump energy consumption.

	Q [m ³ /h]	H [m]	η [%]	Energy consumption [kWh/m ³]
TSS 3.6 g/l	27.2	5.73	59.8	0.0261
TSS 6.2 g/l	26.9	5.75	59	0.0266
TSS 7.4 g/l	26.5	5.77	58.4	0.0269

Table 2. Parameters of the operation points and the specific energy consumption of the pump in case of the three suspended solid contents of fluids

Both the pump performance and the pipeline characteristic curves appeared as expected at high viscous non-Newtonian fluids, especially in the case of TSS 7.4 g/l (see in **Fig. 6**). It should be noted here that if the static head demand of the system increases at several technological points of

the whole activated sludge process (e.g. the water level on the suction side decreases), the characteristic curve of the pipeline may shift upwards. This may result in unstable operation points, which should be avoided from an operational point of view.

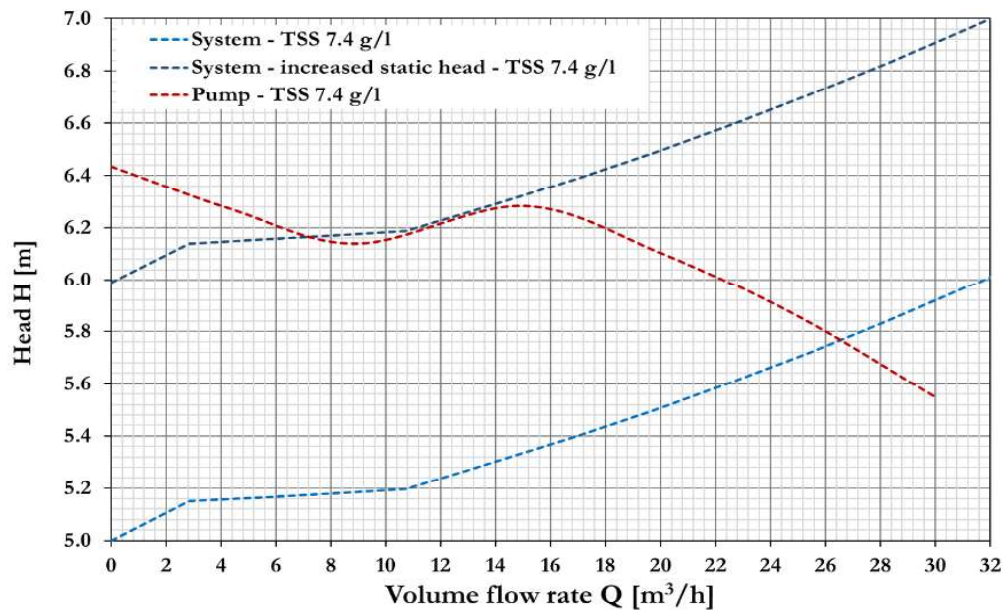


Fig. 6. Operation points in case of increased static head of $H_{st}=6$ m at TSS 7.4 g/l

SUMMARY

Clean water and wastewater technologies play a prominent role in sustainable development. The energy consumption of these essential technologies is significant; thus, the energy-efficient operation is inevitable from a design and operational point of view. The sector's energy use is growing, but its importance is yet under-recognised, so research gaps remain.

This paper indicates that the changes of total suspended solids (TSS) content of the sludge throughout the activated sludge process influence the hydrodynamic conditions and energy consumption of the sludge pumping circle. In addition to the TSS content, the density and the material structure of activated sludge also change during the WWT process. They should also be taken into account in future research.

We have shown that it is possible to reach an unstable operation during the WWT process. Based on this, it can be concluded that activated sludge pumping systems deserve further investigation from the point of view of operation instability as well.

In our study, the rotational speed of the pump was constant. However, a pump system can operate efficiently with frequency converter speed control. Then the characteristic curve of the pump must be transformed into another revolution number using the affinity laws. In the future, the conditions and limitations for the use of

affinity laws should still be examined in the case of non-newtonian fluids.

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NOMENCLATURE

B = parameter described in Eq. (1)
 C_H = head correction factor [1]
 C_Q = volume flow rate correction factor [1]
 C_η = efficiency correction factor [1]
 D = inner pipe diameter [m]
 D_{imp} = impeller diameter [m]
 D_{eq} = equivalent impeller diameter [m]
 f = friction factor [1]
 H = head [m]
 H_{st} = static head [m]
 H' = corrected head [m]
 n = pump revolution number [rpm]
 n' = apparent flow behaviour used in Eq. (7)
 R = radius of curvature [m]
 Q = volume flow rate [m^3/h]
 Q' = corrected volume flow rate [m^3/h]
 v = velocity [m/s]
 v_{eq} = equivalent velocity [m/s]
 w = characteristic dimension of the impeller [1]
 $\dot{\gamma}$ = shear rate [1/s]
 η = efficiency [%]
 η' = corrected efficiency [%]
 μ = dynamic viscosity [Pas]

μ_B = dynamic viscosity of Bingham plastic fluid [Pas]

ν = kinematic viscosity [cSt]

ρ = density [kg/m³]

τ = shear stress [Pa]

τ_0 = yield stress [Pa]

BEP = Best Efficiency Point

CFD = Computational Fluid Dynamics

TSS = Total Suspended Solids

WWT = Wastewater Treatment

Keywords: activated sludge, CFD simulations, energy consumption, hydrodynamic losses, non-Newtonian rheology

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